Practical uses of various data formats

Routine use of varied plots can lead to improvements in machine train operation

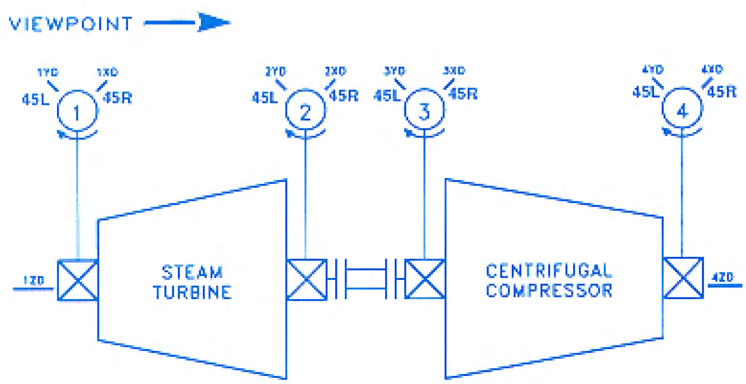

Figure 1
Machine train layout showing transducer locations.

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While performing a typical turbomachinery commissioning vibration analysis, we use many data plot formats to understand and document the machine's behavior. A simple case history from a few years ago illustrates some uses of the various plots.

The machine train was a steam turbine-driven compressor in a hydrogen-rich gas service. The turbine had been extensively rebuilt and fitted with tilting pad-type radial bearings. Bently Nevada's Machinery Diagnostic Services group was hired to monitor the machine's rotor and casing vibrations during startup, so we could promptly determine the cause of any high vibration. We were also to document the unit's dynamic characteristics.

The turbine-compressor train had been fitted with a full complement of proximity transducers (Figure 1. Also see *Proximity Transducer Characteristics* on page 15). Each radial bearing had two Bently Nevada proximity transducers, mounted radially 90° apart in an XY configuration. The transducers were connected to Bently Nevada vibration monitors equipped with alarm relays.

Both the turbine and the compressor were equipped with axially-oriented probes to detect rotor axial movement. These were connected to thrust monitors. A Keyphasor® transducer generated a once-per-revolution pulse to provide a reference for vibration phase measurements.

Recommissioning the machine train

The first step in recommissioning the train was to run the turbine uncoupled from the compressor. We do this to determine the balance state of the turbine, to test the overspeed protection circuits and to characterize the dynamics of the turbine. The uncoupled turbine run revealed low vibration amplitudes at all speeds.

The turbine was shut down for a day, then coupled to the compressor and brought up to the minimum governor speed of 5000 rpm. The unit ran at 5000 rpm for two days while the plant was prepared for the process startup.

We used a Bently Nevada ADRE® 3 multi-channel digital data acquisition instrument and a portable computer to acquire and quickly reduce vibration data. We connected it to the transducers on the warmed up turbine and compressor. Just prior to the process startup, the operators ran the compressor train down to a full stop, while we recorded data.

At low speed, we recorded the shaft slow roll runout. Slow roll runout includes non-dynamic components, such as due to a residual bow in the shaft. The bow, which must be compensated for when balancing, can only be detected by the eddy current, proximity transducer since acceleration and velocity sensors cannot detect slow roll and position data. Because runout can mask dynamic vibration components which are essential for machinery diagnostics, we record it so it can later be used to create runout-compensated data plots.

When the unit came to a stop, we recorded probe gap voltages. We use these as a reference for later estimating the position of the shaft in its bearing. We assume that, with the machine stopped, the rotor is at the "bottom dead center" of its bearings. When the machine is running, the gap voltages change as the shaft journal rises on the bearings' oil film. Given a gap reference for the bottom of the bearing, our diagnostic equipment can plot changes in the shaft's average position while the machine is running.

The turbine-compressor unit had been stopped for only a few minutes when the operators restarted it, using the main hand valve. They intended to bring the turbine up to the minimum governor speed of 5000 rpm. However, the unit accelerated rapidly to 7500 rpm and the monitoring system annunciated an alarm.

Figure 2 is an rpm versus time plot of the brief run. It shows that the turbine was inadvertently accelerated to 7500 rpm and then rapidly slowed to 5000 rpm, where it remained for 20 seconds with a high vibration alarm showing on the monitoring system. The operators then shut the unit down, and its rundown followed the typical speed decay profile.

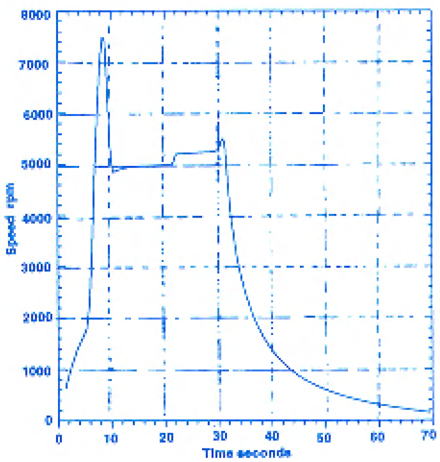

Figure 2
Rpm versus time plot, from abortive startup. These simple, but useful plots are generally used to help explain other data plots and to document overspeed trip tests.

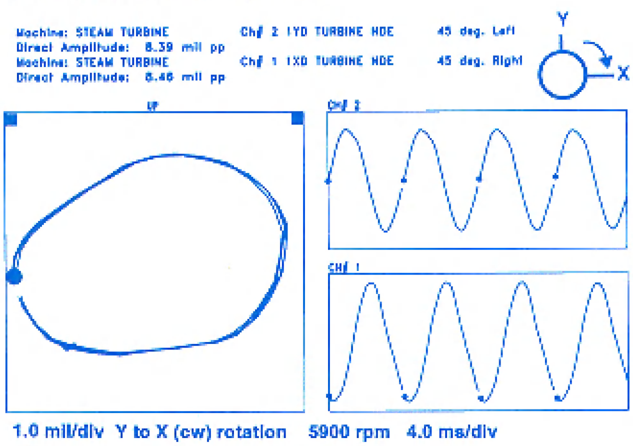

Figure 3
Orbit/timebase plot from the turbine's non-drive bearing, at the time of highest vibration during the abortive startup.
The orbit plot represents the path of the shaft centerline as it vibrates during shaft rotation. An orbit plot is a two-dimensional representation of the orthogonal transducers' ac voltages (vibration signals). The timebase plot is a one-dimensional representation of the instantaneous vibration amplitude as a function of time. It is a paper plot of a standard oscilloscope display.
Orbit/timebase plots provide information on the amplitude of vibration, the direction of radial preloads, the direction of shaft precession, and absolute and relative phase angles.
Precession is the motion of vibration, relative to the direction the shaft rotates, and it helps categorize certain malfunctions. The direction of precession is determined from the timebase plot by noting which transducer, X or Y, that the highest peak of vibration passes first. If, in its normal direction of rotation, an arbitrary point on the shaft would pass that same transducer first, then precession is forward. If not, precession is reverse.

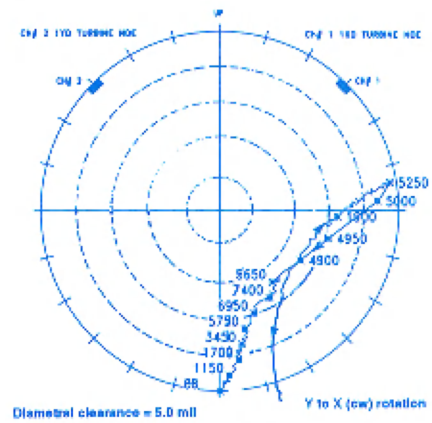

Figure 4
Shaft centerline plot, from the turbine's non-drive bearing, during abortive startup and rundown. The graph's diameter is 127 μm (5 mils), the same as the bearing's diametral clearance.
A shaft centerline plot is a polar representation of the transducers' dc voltages. It shows the shaft's average radial position within its bearing at various speeds. Gap voltages measured while the shaft was at rest are marked at the bottom center of the plot, corresponding to the position the shaft is in while at rest. Gap voltages acquired at various speeds are plotted relative to this point. Shaft centerline plots provide information on coupling and bearing alignment, rotor preloading, bearing oil film thickness, bearing wear and shaft position angle.

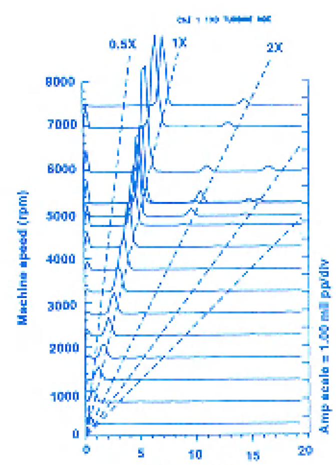

Figure 5
Spectrum cascade plot of rundown during the first startup, from the turbine's non-drive end. This transient data format displays the spectral behavior of the rotor vibration over a range of operating speeds. The plot's vertical axis is the machine speed, and the horizontal axis is the vibration frequency. Diagonal "order of running speed" lines can be drawn to show selected multiples and submultiples of the running speed, in this case 0.5X, 1X, 2X, 3X and 4X the running speed.

Since the unit went into alarm due to high vibration, we were concerned that it might be damaged. The high vibration occurred at the turbine's non-drive end bearing. Figure 3 is an orbit/timebase plot of that bearing's data, at the time of highest vibration. The orbit plot has a circular shape, and the timebase plots show forward precession. On this machine train, with lightweight, high-speed rotors, these features indicate that heavy, rubbing contact between the rotating and stationary parts was unlikely to have occurred. The maximum vibration was approximately 213 μm (8.4 mils) pp; the turbine bearings had only a 127 μm (5 mils) diametral clearance. Had the radial bearings been damaged?

We generated shaft centerline plots, using the gap reference voltages we had just acquired, to see if the bearings had been wiped. Notice, in Figure 4, that at the end of the rundown, the shaft comes to rest within 19 μm (0.75 mils) of its starting position. This is typical of tilting pad bearings with "on pad" loading. Since the bearing temperatures had also been normal, we thought it likely that the bearing was not badly damaged.

How could we have measured more vibration than the bearing clearances would allow? By considering the relative phase responses at each end of the rotor and the locations of the proximity transducers along the rotor, we determined that the rotor was operating in its first, or translational mode. In this mode, the rotor is bow-shaped, with nodes, or points of minimal deflection, at or near the bearings. The proximity probes, mounted several inches outboard of the bearing, measured greater vibration than was present at the bearing (See "Mode Identification Probes" in the June, 1994 issue of the *Orbit*).

The data did not reveal any damage to the unit, so it was restarted. The hot machine train was first rolled at 1000 rpm for three minutes to straighten the rotor, and then ramped up to 5000 rpm. The speed was held at 5300 rpm for three minutes, to complete the warmup. The train was slowly accelerated to 9800 rpm and placed in service without further high vibration incidents.

Although the unit was successfully started, we still wanted to know what caused the initial high vibration. The first step in that analysis was to determine the frequency of the vibration.

We generated a cascade plot from rundown data, after the first coupled startup (Figure 5). It shows that the most significant vibration occurred at 1X running speed. The plot also shows that the unit is stable over the operating range of 0 to 7500 rpm. Since the vibration occurred mainly at 1X, our next step was to examine the transient 1X plots that reveal phase information.

Figure 6 is an uncompensated (not corrected for runout) Bode plot of the abortive startup, generated from turbine drive-end bearing data. It shows the 1X filtered response for both the acceleration and deceleration, and provides a significant amount of information about this vibration problem. Notice that, at low speeds, the amplitude and phase responses for rotor acceleration and deceleration are quite different. This is unusual. At low speeds (less than 5% to 10% of operating speed), the dynamic effects from forces such as unbalance are negligible. The 1X signal at low speeds is the result of electrical and mechanical runout, not actual vibration. This is the runout vector, and it should be consistent from run to run over the life of a particular rotor.

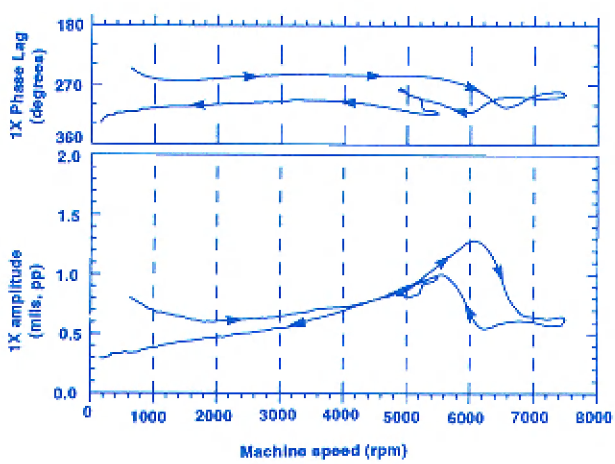

Figure 6

Uncompensated Bode plot of the abortive startup, showing the 1X response for both the acceleration and deceleration (shown by arrows). Bode plots show the amplitude and phase lag response of the rotor-bearing system to the unbalance forces, as the excitation frequency (i.e. rotor speed) changes. The upper portion shows the 1X phase lag. The 1X phase lag is the angle between the unbalance vector (the rotor's "heavy spot") and the 1X vibration (the rotor's "high spot"). The lower portion shows 1X filtered vibration amplitude versus machine speed.

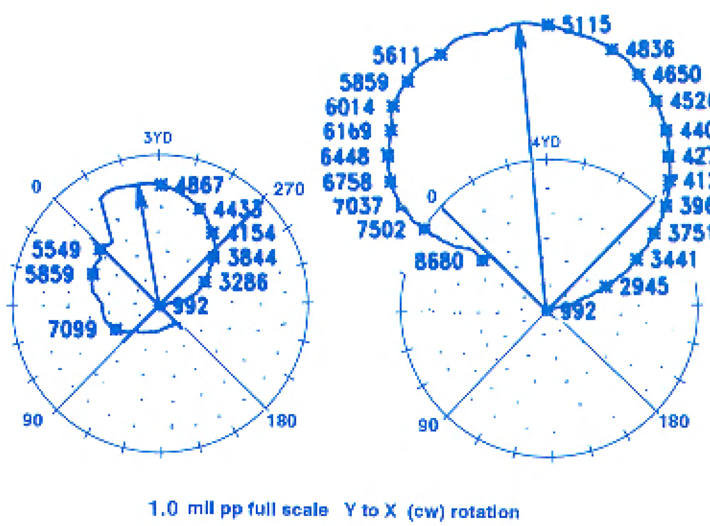

Figure 7

Compensated polar plots from the compressor's drive and non-drive end bearings, respectively, during the second, successful startup. A polar plot contains exactly the same information as a Bode plot; however, some data features are easier to interpret using this format.

Change in the runout vector was also apparent in the data from the turbine's non-drive end bearing (not shown). Although the high vibration had occurred only at the non-drive end of the turbine, matching changes in the runout vectors at either end of the turbine suggested that rotor bow was the problem. When this turbine was run down prior to the first, aborted startup, the rotor was hot, and sat at rest for several minutes before it was restarted. When a hot shaft sits in the bottom of a bearing, its bottom side cools more than its upper side, causing it to bow slightly. A slight, thermally-induced bow in a rotor weighing several hundred pounds can create a high unbalance force at operating speed, with corresponding high vibration amplitudes.

Compressor startup data from the successful second startup gives us further insight into this machine train. In Figure 7 are two compensated 1X polar plots: on the left from the compressor's drive end, and on the right, from its non-drive end. As speed increases, corresponding data points on each plot follow a similar path, indicating that the 1X vectors are generally in-phase. Maximum vibration amplitude occurred at approximately 5300 rpm, accompanied by a 90° phase shift. These are all characteristic of the first balance resonance. At 8680 rpm, the compressor operates well above its first balance resonance.

During the second startup, the unit had been run at 5300 rpm for approximately three minutes. Notice in Figure 7 that the drive end (left plot) 1X vector decreased in amplitude in a stepwise fashion at approximately 5363 rpm. The decrease in vibration was likely due to a decrease in the thermally-induced turbine rotor bow, transmitted through the coupling to the compressor. This, and the evidence contained in the spectrum cascade and Bode plots, indicate that the slow roll period prior to runup was not sufficient to remove the thermal bow. Therefore, our first recommendation was that the slow roll period for a "hot" start be increased from three to fifteen minutes.

The high speed soak, at 5300 rpm, is at the first balance resonance of the compressor rotor. The compressor and turbine appeared to be well-balanced during these tests. If the compressor rotor had not been well-balanced, higher vibration amplitudes caused by sustained operation at the first balance resonance (5300 rpm) would probably have damaged the compressor labyrinth seals. Therefore, our second recommendation was to change the high speed soak speed to 3500 to 4000 rpm, to limit vibration amplitudes during this phase of a normal startup.

Conclusion

No single data presentation format fully characterizes machinery behavior. Different formats emphasize different characteristics. Using each data format appropriately makes the analysis of vibration more accurate and effective. ■

Proximity Transducer Characteristics

The effectiveness of turbomachinery protection systems over the past thirty years owes a great deal to the availability of non-contacting, eddy current proximity transducers. Developed into a practical industrial device by Donald Bently in the mid-1950's, these transducers are used to measure the vibration and position of a machine's rotating parts with respect to its stationary parts.

Unlike seismic transducers, such as velocity transducers and accelerometers, which indirectly indicate rotor vibration, a proximity probe measures rotor position directly. Its output has two components. One is a direct current (dc) level that is proportional to the average shaft position. The other is an alternating current (ac) signal proportional to the instantaneous shaft position.

Figure 1 shows a typical eddy current transducer. The operating end of the transducer usually takes the form of a small threaded rod, which is secured to the bearing cap and adjusted to measure the movement of the rotating shaft relative to the bearing. On critical machines, they are installed in orthogonal pairs, so the motion of the shaft in the plane of the probes can be fully measured.

Figure 2 shows an eddy current transducer system calibration curve for a typical vibration transducer. The proximity probe is connected to its oscillator/demodulator (Proximitor®) module by coaxial cable. The system must be calibrated to the specific material the probe observes, usually AISI 4140 steel. The usual sensitivity is 7.87 mV/μm (200 mv/mil). The frequency response of the transducer system is dc to 10 kHz, and its total linear range is approximately 2.54 mm (100 mils). The American Petroleum Institute publishes API Standard 670 to set common design criteria for eddy current displacement transducers and the monitoring systems to which they are connected. ■

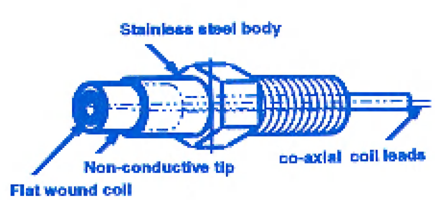

Figure 1

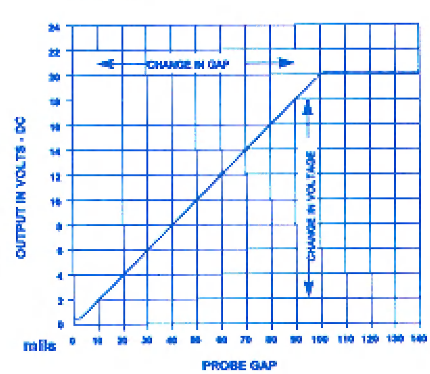

Figure 2